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LONG LIFE BALL-BEARINGS FOR USE IN SATELLITES IN SEALED CONTAINERS

by

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SUMMARY

The Report concerns the development of long life ball bearings with one initial lubrication. The knowledge of many experts was pooled and an experimental batch of 60 spindles fitted with several types of ball races was run for seven years. The conclusion is that a 10 year life should be reliably obtained for such devices as momentum or reaction wheels for satellite attitude control. Significantly longer reliable lives are probably possible.

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1 INTRODUCTION

In 1961 Space Department initiated the development of a control moment gyroscope for satellite attitude control purposes which was required to have a 5 year life. At various times since 1941 the author had been concerned with either the manufacture or the development of ball races for special purposes and had many contacts in industry. Inquiries were therefore made to firms in this country as to the reliability of ball-bearing spindles run continuously and having one initial filling of lubricant. Some data were obtained on a small number of marine gyro spindles which had run for periods of 2 or 3 years and one marine gyro compass which had run for 6 years, without re-lubrication in both cases, but, in general, information was lacking. Further inquiries were made during a visit to the USA in 1961 concerning momentum exchange devices. It appeared that no one had made batch life tests of ball races with one initial filling of lubricant. Obviously there was need for work to be done in this area.

The reliability and life data given in bearing catalogues relate to metal fatigue on the assumption that the bearing is periodically re-lubricated. In the weightless environment of space, loadings can be made so low that fatigue is not a problem, but extensive lubrication systems are impracticable. The problem of long term reliability for space use is largely one of the lubricant and the effect of the physical parameters, the environment and mechanical design upon the lubricant.

The lack of information on long life ball races led to a decision to use a self-generating gas bearing for the gyro spin axis and also that some experimental work on long life ball-bearings should be started as an insurance policy.

This Report concerns the philosophy that governed the design of 60 ball bearing test rigs and a description of the test results after approximately 7 years of running for most of the spindles. The size of bearings was appropriate to the control moment gyro, but the philosophy and test results have a more widespread application, e.g. momentum and reaction wheels.

The concept of the experiments made use of the combined knowledge, as it was in 1962/3, of the author, two gyro firms, IAP Department of the RAE and the laboratory staff of a major oil company. The choice of bearing types was dependent on and subsequent to an appraisal of this combined information and has been put in that order in this Report. However, it may help the reader to look at Table 1, which lists the bearings used, before reading further.

2 ENGINEERING DETAILS FAVOURABLE TO LONG LIFE

The results of the inquiries mentioned in the introduction are given here. Each detail is important because in bearings there is a tendency for one bad effect to adversely react on another parameter to make the situation worse. For example, a mechanical error in assembly can produce a high load, which results in a high wear rate. The high load also increases the temperature which increases the rate of chemical degradation of the lubricant and its evaporation rate, and reduces its viscosity. This in turn produces a thinner oil film and increased wear rate. It is easy to see how such effects can accumulate to cause a failure.

2.1 Lubrication

In 1962/3 none of the people consulted had experience of elasto-hydrodynamic lubrication[1] and this was not taken into consideration. It should, however, be taken into account in future design work.

Inertial navigation gyro bearings were oil lubricated, but some information of gyros for ships and military vehicles indicated that grease gave a longer life. This accorded with advice from oil and grease manufacturers. The author's own experience of 12.7 × 28.6 × 6.35mm wide ball races with strip metal cages, which were used in a development version of a 24000rev/min turbo alternator was that bearings lubricated with oil lasted only a few hours, but when an equivalent quantity of grease was used no failures occurred in tests of up to 2000h duration.

Oil migration by creep was known to be a hazard during storage or running time. Experiments with the bearings for the alternator showed that liquid oil from the grease could be seen on the balls after 24 hours running. The appearance was much the same after 2000 hours and a ring of grease on the lands (principally the outer race) appeared to prevent this creeping away. The metal cages of these bearings suffered severe adhesive and abrasive wear if they touched the lands, that is, ceased to be ball riding. Non-metallic cages were indicated and this fitted with the need for ample lubricant storage.

Considering that extreme long life was the aim, some store of lubricant for replacing oil lost by evaporation or creep was desirable. For simplicity of design, this store could only be in the bearing.

For the majority of bearings it was therefore decided to use porous nylon ball cages, which were impregnated with the oil used in the grease and, in addition, a small amount of grease was smeared on the inner race and was spread around by running. The lubrication was therefore principally oil plus a little grease. Some bearings had non porous cages and here only grease was used.

The choice of lubricant could only be sensibly made as a result of previous experience. The Sperry Gyro Co., Bracknell, provided the results of life tests with a number of greases and recommended Aeroshell 12, which had done well in general and, in addition, had given 6 years continuous running in a marine gyro in a laboratory test. The Shell Laboratory staff agreed with this choice, because of this experience, but would not have done otherwise. This grease is composed of a specially waterproofed silica in the base oil di(2 ethyl hexyl) sebacate, with a small amount of an amine anti-oxidant. It is a high temperature range lubricant, -54°C to +200°C, to meet DTD 900/4222. It is not now available commercially.

2.2 Low lubricant temperature

Despite the high temperature range of this grease, the Shell Laboratories emphasized that where long life is the aim the lubricant temperature must be kept low, to minimise the rate of chemical degradation. This is a general rule for any lubricant. The author's previous experience had confirmed this and subsequent experience in the gyro field showed that it is vitally important. In addition, the appearance of the substance known as 'friction polymer*', which is supposed to be due to running and not just time alone, is very temperature dependent. Furthermore, the evaporation loss rate of an oil varies with vapour pressure and this increases by an order of magnitude for each 20°C rise of temperature. A top temperature limit of 50°C was chosen therefore and this has been confirmed as being reasonable by subsequent experience, assuming the particular lubricant to be of good chemical stability, e.g. one of the standard gyro mineral oils. Clearly, the lowest attainable temperature is desirable if extremely long lives are required. This temperature limit applies to the lubricant itself and any temperature gradient between it and the position where the temperature is measured must be considered.

2.3 Ambient atmosphere

Lubricating oils do oxidise, albeit slowly, at normal temperatures. A thin film of oil in a ball race has a very large surface area to mass ratio, which increases the rate of deterioration. One of the panel of advisers had discovered

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* 'Friction polymer', is the name given to a brown deposit sometimes found at the side of the ball track and usually thought to consist of polymerised lubricant mixed with wear particles from the cage and raceways.

experimentally that the life of gyro bearings doubled when sealed in a helium atmosphere. Helium is the standard fill for encapsulated gyros, because with it windage losses are low and heat transmission is high. However, there are certain disadvantages in using very pure helium. The Shell Laboratories had shown[2] that very pure helium resulted in surface distress on the ball tracks at an early stage. Further, some boundary lubricants need oxygen and/or water in order to function and this is also true for the oxide on the metal, for protection against adhesive wear. Therefore it was arranged to run the experimental bearings in an atmosphere of helium with 5% of air, i.e. the mixture would contain 1% of oxygen and a little water vapour.

2.4 Quantity of lubricant

The quantity of lubricant in inertial navigation gyros is limited to a few milligrams, because of the adverse effect of mass migration on wander rate, particularly during acceleration. For maximising life this was considered to be a bad approach. To achieve maximum life one grease manufacturer recommended filling the bearing to capacity and allowing the surplus to extrude during initial running. This did not appear to be very suitable for the control moment gyro, as grease would get to unwanted places, but the idea was tested and it was found that severe ball skidding occurred during the initial running phase.

Conversely, various authorities recommended filling the bearings to one third capacity. It was also the common opinion that a quantity of oil which was more than that normally used for a gyro, but which was much less than that employed in a fill for industrial purposes, was the right approach. This intermediate amount could be used in a control moment gyro, which employed ball bearings, provided that ground testing was conducted with the output axis vertical. In the event, the cages were fully impregnated with oil, amounting to 40-60mg, and about 10mg of grease were smeared on the inner track, to be spread around by running. Table 1 shows the average fill for each group. By some mistake, group E cages were centrifuged down to 14mg of oil, which at least provided another interesting variant.

For reasons discussed later, two types of bearings with non-porous cages were added to the test programme. These were filled to one third capacity with grease.

2.5 Inner race rotation

The relative velocities of the moving parts of a bearing are independent of which race is rotating. However, depending on the actual value of the speed, there can be a significant advantage for the lubrication process if the inner race is made the rotating member. The centrifugal forces in high speed bearings causes the lubricant to migrate to the outer race. The author's experience has shown that oil tends to be thrown out of the bearing altogether, but grease forms a ridge on the outer race lands which helps to retain the oil. Depending on speed, retention is more effective if the outer race is stationary. For the bearings of 16mm outer diameter used for these tests, the centrifugal force on the outer race lands would be 1040g if these were rotating at 12000rev/min - the higher of the two test speeds.

Another effect occurs in small bearings which favours inner race rotation. For example, in the case of the size chosen for test, the cage angular velocity for inner race rotation is about half what it would be for outer race rotation. Therefore, at 12000rev/min, there would be 360 or 96g applied to the cage, depending on the choice of the rotating member.

The above reasons indicated that, in the test speed range of 6000 to 12000rev/min, inner race rotation was the best choice for long life and so it was arranged for the outer race to be fixed.

Incidentally, at the time of making this decision there were no published data on this topic: some information is now available, but it deals with grease only. One shot lubrication with oil is something commercial lubricant firms consider to be suitable only for slow moving instrument bearings, possibly because oil-absorbent cages are not in ordinary use.

2.6 Bearing sealing

Fig.1 shows that the rotor of the test rig would act as a centrifugal pump, drawing gas through the bearings, unless these were sealed. The panel of consultants considered that such a flow of gas would be undesirable, because of the possibility of dirt being carried in and also of some increase in oil evaporation rate. Therefore the bearing enclosures were sealed as shown in the figure. Previous experience with an 0.36m diameter disc running at 2400rev/min is of interest. Before this fault was recognised and cured, the passage of air through the bearings was such as to extract the grease and spatter it on to the transparent cover of the rig.

2.7 Fitting accuracy

It was common knowledge that many bearing failures had been caused by bad fitting, and care was taken to avoid this. Precision ball races are ground on precision machinery and it often happens that the machinery used for the shaft and housing seating surfaces is not of equivalent accuracy. The bearings used in these tests were better than ABEC 5 grade and this implies roundness accuracy of better than 1μm on the bearing mounting surfaces and ball grooves. The effect of errors in roundness is to concentrate the axial loading, which should be evenly distributed around the races, into local areas. Fatigue life varies inversely with the cube of the load, and high load areas could also develop high local temperatures, with detrimental effects on the lubricant. Interference fits of miniature bearing rings force the rings into almost exactly the shapes of the mating surfaces and can alter the contact angles of angular contact bearings or the diametral clearances of deep grove types. To avoid any possibility of such distortions occurring, no interference fits were used. Instead three rings in each assembly were clamped axially and the fourth was a sliding fit and was axially loaded by the preload mechanism. The bearing diameters were measured and the particular batches were found to be of the same dimensions within much less than the allowable tolerances, which helped in obtaining individual fittings. The housings and shafts were ground to these dimensions to better than 2.5μm and in some tight cases were finished by lapping, so that a smooth sliding fit was obtained in all cases.

2.8 Angular misalignment between inner and outer races

Ref.3 is a collection of five RAE reports all of which deal mainly with the problem of angular misalignment between inner and outer races. The effect of such misalignment is to cause the balls to have different orbital velocities at different positions around the race, thus some are pushing the cage in the right direction and some doing the opposite. Depending on the magnitude of the misalignment and of the axial preload, which tends to prevent the balls skidding to relieve the force, the resulting forces on the cage can be large. In one extreme case quoted in the above papers, all the cage rivets broke in tension. Ref.4 gives an example of a 10mm id 26mm od bearing with 4.76mm (3/16in) diameter balls where, for a misalignment of 0.002mm/mm, the ball travel from its mean position is quoted as being ±0.23mm (±0.009in) (cage permitting) and contact angle variation ±7.5° about a mean of 15°.

Apart from errors in the bearing itself, this misalignment can be caused in two ways. The first is that the inner or outer seatings have errors of parallelism with the axis of rotation. The second is that, the housings at the opposite ends of the shaft do not have a common axis. It was clearly desirable to minimise these effects. The outer race housings were therefore bored straight through in one operation, the removable pillar (Fig.1) being screwed and dowelled into position for this operation. As the shaft was one piece, which could be reversed in the chuck, good alignment here was probable. Fig.1 shows that the axial clamping system is of good inherent accuracy.

2.9 Preload and stiffness of preload spring

It is common observation that ball bearings which are lightly loaded, such as small fans, refrigerator motors, etc., have long life times, frequently running many years on one initial fill of lubricant. Small motor bearings tend to be large in load carrying capacity relative to the applied load. One reason for this is that any reasonable diameter of shaft plus the thickness of the inner ring of the ball race tends to turn out this way. The conventional meaning of load in a ball race is in reference to fatigue life. However, it is clear that what is good for fatigue life is also good for lubricant life. The reasons are that light load is conducive to low temperature with beneficial effects, as discussed in section 2.2, and low wear particle production, which also is good for the lubricant. Low load is also necessary for fatigue life, but, as mentioned in section 1, this should not be the deciding factor. In the subsequent experiments the fatigue life was made in excess of 300 years.

Gyro bearings are the opposite to the above in design. The bearings are heavily preloaded in order to make them stiff and the preload spring is also stiff, being just the stiffness of the shaft and housing. Typical axial loads are 9N to 18N on R 1.5 to R 3 size bearings 2.38 to 4.76mm (0.093 to 0.188in) bore size and rotor weight is about 20 grams. Hertz stress approaches $2.0 \times 10^3 MN/m^2$ (300000psi) in one case. This high preloading is used to stiffen the assembly and so minimise mass unbalance developed during acceleration, which would result in gyro pointing error. This arrangement demands great accuracy and good temperature control, otherwise large spurious loads are produced in the bearings, with ill results as mentioned in the previous section. In general, small high accuracy gyros in standard form had bearing life times of about 3 months, at the time of designing the experiments reported here, 40 times less than that required for a communication satellite.

As an illustration of the effect of mechanical errors where the preload is solid, being produced by the compliance of the shaft, housing and bearing only, one might consider the AEEC 5 track wobble limit in 1962/3 of 7μm (0.0003in). Extending or compressing the shaft by this amount produces a change of load of approximately 2000N (450 lb), the housing is of course much stiffer. Assuming that both bearings are on their adverse top limit, compressing the bearing balls into the raceways by this amount would produce a change of 45N (10 lb) which still varies the preload by an order of magnitude.

Some experiments conducted by the Sperry Gyro Co. further illustrate this point. In the 1950s six gyros were used in a test to discover the effect of compliance in the preload system on life. Three were left with their normal solid preload and three were given a soft preload spring. This modification increased the average life by a factor of 6.

Therefore, from consideration of the above information, the 5mm bore bearings under test (which were primarily intended for a gyro where mass unbalance was no problem) were given a low preload of 4.5N and a compliant preload spring, so that this value would not be changed significantly by mechanical errors.

2.10 Material and grade of bearings

The choice of bearing material, then as now, lay between corrosion resistant steel, about 12% Cr 0.5% C, and the conventional 1% Cr 1% C ball-bearing steel. In the space application of these bearings, it was thought that they would run in an atmosphere of some inert gas and there would be no corrosion problem. Therefore it seemed worthwhile to take advantage of the greater fatigue life of the normal steel. Furthermore, stainless steel bearings did not have much user experience behind them at that time. After the test, a sample of each type was analysed and confirmed to be either EN 31 or 52100 - the normal bearing steel, British or American version.

Precision grade bearings were used. As well as finer tolerances, the selection of this grade ensured a much greater degree of quality control and the price was about an order greater than for the commercial grade. Nearly all the bearings were supplied by a single company, but some had been made in the UK and some imported from the USA. The best available accuracy to which all the types used could be obtained was ABEC 5 or "slightly better than ABEC 5". ABEC stands for the Annular Bearing Engineers Committee of the Anti-Friction Bearing Manufacturers Association Inc. (AFBMA). ABEC 5 is the lowest of the precision grades, but it is substantially finer than the commercial grade. Bearings with much finer tolerances are now available.

3 RIG DESIGN

Much has been stated on this subject already and Figs.1, 2 and 3 almost complete the picture. These rigs still exist, and may be used again.

The baseplate, which includes the motor stator housing and one bearing housing, is an aluminium alloy casting. The other bearing housing is machined from aluminium alloy bar and it is dowelled and screwed to the base. The bearing outer races are contained in austenitic stainless steel cylinders of substantial radial thickness (3.2mm) which are interference fitted into the aluminium. A stainless steel plunger is used in one cylinder to transmit the spring preload to the outer race of one bearing. This choice of like materials was a mistake, because in a few instances sufficient fretting occurred between the plunger and the cylinder to cause seizure, despite plentiful lubrication. A Bell jar is clamped to the base and sealed by an O ring, and a Schrader valve is provided for gas filling and sealing.

A 400Hz synchronous motor of the hysteresis type is used to drive the shaft, with the hysteresis ring glued to the mass rotor with epoxy resin. The rotor shaft is steel and therefore there is an appreciable thermal expansion difference between shaft and base. This was arranged deliberately because it represents a situation which satellite bearings must tolerate. The $20^{o}C$ temperature cycling applied in the tests resulted in 11μm of relative movement between one outer race, with its preload plunger, and the housing cylinder. The rig temperature is sensed by a thermocouple which is embedded into the motor housing close to one bearing.

4 CHOICE OF TYPES OF BEARING

The choice of bearings hinged upon the need to have a reasonable size from general engineering considerations, including fatigue, and also the requirement that a number of variants should be available in that size. It was common experience, and still is, that very small bearings, say 3mm bore and downwards, frequently do not give the life that theoretically one might expect. The precise reasons for this do not appear to be known, but one is probably because they are difficult to make; the tolerances are fixed for a large range of diameters, so that, proportionally, manufacturing errors are larger in small bearings.

The rotor weight was 165g and a reasonable shaft diameter (5mm) coincided with a range of bearings available from a single manufacturer, all of 5mm bore, 16mm outer diameter and, all but one, 5mm wide. All have six or seven English size balls, 3.17mm (0.125in) diameter. It is reasonable to expect bearings of this size to be manufactured to a good standard.

The fatigue life was calculated from the manufacturer's data using the minimum contact angle for the AC bearings of 17° (maximum resultant load) and 4.5N axial preload. The result was 350 years at the maximum test speed of 12000rev/min. This is the conventional L10 life defined as the minimum life for 90% of a typical group of apparently identical bearings. For satellite use a reliability of 0.99 is necessary and for this the life is 70 years. Apparently there was enough fatigue life available to remove this factor from the list of problems. In fact the tests showed that there was some fatigue in the extra wide deep groove bearings, as will be described later.

The types of bearing tested are shown in Table 1 and the reasons for the various choices and descriptions of the bearings are as follows:-

Groups A and D: Type 1 (Fig.3). 5mm × 16mm × 5mm. Integral inner race. Seven balls of 3.17mm diameter. One piece sintered nylon cage ('Nylasint'). Contact angle range 17° to 27°. Preload 450g. This was a gyro bearing made with the inner race as an integral part of the shaft. At the time of its selection, this design was recommended by Avionics Department, because it eliminates one joint per bearing and, on average, there is therefore some improvement in the alignment of the inner race track. This feature is clearly more important in solid preloaded cases, such as gyros, than for the bearing installation used in these tests. However, it was still considered to be worth including.

Groups B and E: Type 2 (Fig.4). 5mm × 16mm × 5mm. This is similar to Type 1, except that the inner race is not part of the shaft. It is a conventional bearing and was the obvious first choice by which the others would be judged.

Groups C and F: Type 3 (Fig.5). 5mm × 16mm × 5mm, and Type 4, 5mm × 16mm × 6mm, which is similar. Shielded type deep groove bearing used with the bearing shield removed, six balls, cage of corrosion resistant steel strip. Diametral clearance 7.5μm to 12.5μm. These bearings were used without preload in the conventional manner for this type, i.e. one end was fixed and the other free to slide axially. These bearings were included because, by the author's observation, they perform so well in small motors, despite the text book homily that preload must be used to prevent ball skid. In the event they did well. The diametral clearance is one grade higher than the manufacturers' recommendation because the author's previous experiments had shown that a high clearance tends to reduce the power requirement and temperature. Since the cage was not porous, the lubricant was grease only.

Group G: Type 5 (Fig.6). 5mm × 16mm × 10mm wide. Deep groove extra wide series. Six balls, riveted phenolic cage with aluminium alloy side plates to the cage. The idea of producing a series of bearings identical in their working parts to normal, but of double width, was a move towards greater reliability. This range of bearings in the normal width can be criticised on the grounds of narrowness in relation to diameter, which mitigates against good alignment and also contains very little lubricant space within the bearing. While alignment had received special attention in our tests by use of axial clamping, which made use of the diameter rather than the width as the alignment datum, advantage was taken of a group of 16 of these bearings being available to include them in the tests. At that time they were only made to special order and this group was a production surplus. The phenolic cage was desirable as a variant, as it is the common high speed material for cages. In the particular application known of in the USA, similar bearings were used as AC preloaded bearings and this practice was followed in these tests. It was not realised at the time what a small contact angle these bearings had when used in this mode, with consequent high resultant ball load. By measurement, the contact angle proved to be 6° in one case. It was these bearings which showed some fatigue damage. The cages were not porous, so the lubricant was grease only, and a large quantity was inserted.

5 OTHER TEST PARAMETERS

5.1 Number of spindles

The scatter on fatigue life of commercial bearings was known to be about 10:1 and lubrication life was thought to be much the same. Advice was sought from Mathematics Department, RAE concerning the minimum number of spindles in each group to get meaningful results. The agreed number was 10, i.e. 20 bearings. The total number of spindles was 60.

5.2 Speed

The speeds chosen took into account solar cell output per unit weight. The idea was that the total weight of the damping gyro and its power supply should be minimized. The faster the wheel speed, the smaller and lighter was the gyro for a given momentum, but extra power and consequently weight of solar cells was required. Initially the optimum speed was 6000rev/min, but a year later, because of improvements in solar cell technology it had risen to 12000rev/min. Both speeds were used.

5.3 Temperature

Arrangements were made to vary the temperature cyclically between 25° and 45°C over a 90 minute period, because it was thought at that time (quite wrongly) that this would occur to a component inside a satellite in near earth orbit. However, temperature changes do occur, if not of this magnitude or so frequently, and since the bearings performed very well, this overtest did some good in terms of providing additional confidence.

5.4 Radiation

The Shell Co., had not done radiation tests on the proposed lubricant and, although they thought it would not be harmful, they recommended that such tests be included. Incidentally, they had done many irradiation tests for the nuclear power industry and quoted a rough figure of 1000 megarads to the point where mineral oil degraded. However, Aeroshell 12 is an ester.

It was therefore decided to lubricate 5 out of each batch of 10 spindles with lubricant which had been previously irradicated with 3 megarads of gamma radiation. This, unfortunately from statistical aspects, reduced the number of spindles per batch, but because of cost there was no option, and, anyway, the radiation effects were not expected to be significant. This dosage represents three years in the Van Allen belt.

5.5 Track wobble

IAP Department RAE had developed "A bearing quality measuring machine"[5] which measured the error in orthogonality of the inner race of each of two bearings on a spindle with their common axis. This error is known as track wobble or weave. It also measured the total axial movement of the spindle due to the wobble errors in both bearings. It should be noted that, where the inner race is not an integral part of the shaft, an error of this type occurs, due to the extra joint between the inner race and shaft. The machine was primarily intended for gyro bearings which are solidly preloaded and where, therefore, these errors can produce large changes in load. Ref.5 gives examples. With a soft preload such as was used for these tests, this problem was apparently overcome. However, as the test spindles and bearings would fit the machine, the track wobble of all the bearings tested was measured and an attempt was made to correlate the results with track damage at the end of the test.

6 RESULTS

6.1 Procedure

When the 6000rev/min bearings had reached 61000 hours and the 12000rev/min ones 54000 hours, it was decided to stop the test and examine all the bearings. Swansea Tribology Centre (STC) had been given a contract to assist in this.

The rigs were dismantled at RAE and a large number of the separable bearings were examined using a binocular microscope arranged at X22 power, before passing them to STC. For this examination, the inner races were placed, with little disturbance to the lubricant, on a spindle made to fit the bearings and which would be rotated smoothly between ground centres.

Swansea Tribology Centre had contracted to:-

Examine all bearings with a hand microscope.

Extract the lubricant from the bearings and measure its quantity.

Extract the oil from the porous cages and measure its quantity.

Examine the cleaned bearings with a high power optical microscope and take photographs of the tracks. This involved dismantling the cages of the deep groove bearings.

Use a Scanning Electron Microscope where required.

Make metallographic sections where these seemed to be desirable.

Make track roundness measurements using a Talyrond.

STC wrote a report on this investigation which included a description sheet for each bearing and some very good photographs. The clean bearings were returned to the RAE where several microscopic examinations have subsequently been made of all of them.

6.2 Summary of results

The reader should refer to Table 2 for 'quick look' results.

Twenty-nine out of the 30 6000rev/min spindles survived the 61000 hours test and most of them were capable of several more years running. Only 17 of the 30 12000rev/min spindles finished 54000 hours; approximately half of the failures occurred in the final 2000 hours of running and the other half were rig failures. This sharp incidence of failures at the end of the period showed that all the 12000rev/min groups were at the end of their useful life, but the reasons for this varied between the different groups.

The one failure in the 6000rev/min groups, an integral inner type, which occurred at 33000 hours, was a bearing failure, but it was not in any way typical and appears to have been due to human error in initial assembly. The running torque of this spindle increased from the beginning of the test and STC could find almost no lubricant in one bearing. STC estimated the remaining average life for the 19 other bearings in groups A and B (ac preloaded, 6000rev/min) to be an extra 2.5 years, making 9.5 years total when added to the seven years already run. RAE consider that this estimate was conservative.

The initial examination at RAE after the tests was very interesting as regards the behaviour of the grease. In the case of the 6000rev/min spindles, the grease had formed ridges on either side of the ball track and a quantity of oil was retained between these ridges. The grease had formed a safe anti-creep barrier retaining the oil where it was most needed. The 12000rev/min bearings did not show this effect so markedly, the centrifugal force of 600g had largely thrown the grease off the inner races.

The condition of most of the tracks of bearing in groups A and B was so good that no difference could be seen with the naked eye between them and a new bearing. Where there was a difference, it was a dirty stain, not damage. The STC report states "These groups as a whole showed very little wear of the bearing elements, not enough, in most cases, to completely remove the original surface finish". It is also stated that in most cases an electron microscope was necessary to identify individual effects. This is surprising, because the oil film thickness was insufficient to prevent contact, except at low temperatures, as Fig.7 shows. The elasto hydrodynamic film thickness was calculated subsequently to the tests and also checked experimentally. The boundaries between the lubrication regimes in Fig.7 were derived from averaging the surface finish across the tracks of 8 bearings. These measurements were made after the tests and there was a factor of 3 difference between the best and the worst, which illustrates the approximate nature of this aspect of lubrication.

The tracks of group C bearings (no preload, deep groove, strip stainless steel cage) were very well marked, worn and wavy. Much ball skidding obviously had occurred. The grease was black in appearance and therefore probably full of iron oxide. However, all 10 spindles completed 7 years and would have done more, which shows the reliability of this very simple system.

Groups D and E, the 12000rev/min duplicates of A and B, had so many rig failures that these experiments approached being spoiled. However, some valuable results were obtained which are described in the relevant sections. Failures which can be identified as being caused by genuine bearing troubles were apparently due to lubricant failure, which lead to cage breakages.

The best performers at 12000rev/min, without any doubt, were the double width phenolic cage deep groove bearings of group G, which were preloaded and had an enormous lubricant fill of 210mg of grease (no oil). Seven spindles out of 8 survived the test. All tracks showed some fatigue, one bearing very badly, with cracks in the race way and in a ball, but this is to be expected, as explained below.

When the contact angle of one bearing was measured at the end of test, it turned out to be 6°, much less than it was originally thought to be and, by observation of ball track position, it appears to be representative of the others. In fact, some contact angles may have been lower. Hence the resultant ball load was some three times greater than that of the other angular contact bearings and the corresponding fatigue life would be less than those others by the cube of this factor, i.e. 27. The 350 year L10 life accordingly is reduced to 13.0. This is near enough to the test period for fatigue effects to show.

The two group F spindles, supported on deep groove bearings which ran at 12000rev/min without preload, ended up in dreadful condition, full of material from fretting corrosion. The space next to each bearing was also filled with about 1 ml of these products.

RAE and Swansea Tribology Centre agree that the group which was in the best condition overall was group B, the detachable inner, 6000rev/min 4.5N preload, angular contact bearings, with sintered nylon cages. Theoretically these had a spread of contact angle from 17 to 27°. All bearings finished the test and there were no rig troubles, such as the preload plunger seizing, which bedevilled the 12000rev/min rigs. It was therefore possible to observe that the track condition was the best, being marginally better than group A, without confusion because of rig failures.

6.3 Effect of track wobble

Track wobble varied considerably from bearing to bearing, but comparison of the measured values with the corresponding track condition at the end of the tests showed no correlation. For example the average track wobble for group A bearings, in the units in which the machine measures, were 0.683×10^{-3}in at one inch radius and for group B 1.043×10^{-3}in. Group B bearings had therefore wobble errors nearly twice that of group A (which is caused by the joint between the inner race and the shaft) and yet group B bearings performed better. It was therefore concluded that track wobble had no significant effect and this is one justification for using a compliant preloading arrangement, ability to absorb expansion mismatch is the other.

A condensed version of the track wobble values in microns at the radius of the outer race is given below. They are for inner race rotation as given by the IAP machine[5] and include shaft errors.

Bearing group	A	B	D	E
Worst sample	14.0	19.0	14.5	10.0
Best sample	0.9	1.2	0.6	1.0
Mean of 20 bearings	5.5	8.4	2.3	3.6

6.4 Effect of irradiation of the lubricant

No difference could be observed between bearings with normal and irradiated lubricant.

6.5 Effect of lubricant quantity

All the bearings in group E had cages which had been centrifuged after impregnation to reduce the oil quantity to a mean of 14mg, as compared with 55mg for group D which also ran at 12000rev/min. Unfortunately these were the groups which experienced rig troubles. However, the surviving 8 bearings from D were rated by STC and RAE as 5 good, 2 fair and 1 poor; while from E, 5 were good and 3 poor. So, superficially, there was nothing to choose between them. However STC, in their 'visual comments', frequently say 'clean dry bearing' in group E and not so for D. Consequently nothing was lost by leaving the cages full of lubricant and in fact there was an advantageous side effect, inasmuch as there were fewer cases of fretting in the housing and consequent preload plunger seizures. Therefore it would seem best to use the greater quantity of lubricant, unless it is clear that damage may be done by migrating lubricant. Possibly a lubricant fill in between 55 and 14mg would be a better choice, since it is said that a completely full cage loses a lot of lubricant in the first few days of running.

6.6 Effect of speed

Groups A and B were identical bearings to D and E, the speed being 6000rev/min in the former groups and 12000 in the latter. From this it had been hoped to assess the effect of speed. While there were no rig failures in the 6000rev/min, there were many in the 12000. However, in spite of these unfortunate circumstances, some comparison is still possible.

In groups D and E the test duration was 54500 hours and there were 8 surviving spindles, 4 in each group. Some of these were in dry condition, others had cracked cages and therefore, in general, their remaining life expectancy was small. There were 4 bearing failures, 3 in D and one in E, which occurred within the last 2000 hours and all appeared to be lubricant failures, with signs of heat and black soot - like debris everywhere. There was one early bearing failure which had the signs of being due to a human error, and is therefore best ignored. The remaining 7 failures were all due to rig troubles which had nothing to do with the bearings. The 12000rev/min rigs were made 6 months later than the others and the quality may have deteriorated, as they were made to a limited budget.

From the above list it is obvious that groups D and E had reached the end of their life. Reasonable reliability must be applied to the term life, so this must certainly be less than that of the 4 bearing failures. A reliable life of 10% less than this, 47250 hours or 5.4 years is suggested for D and E, assuming that rig failures be discounted.

By both the author and STC's assessment, the 6000rev/min groups had appreciable remaining life after 7 years test running and STC gave a rough estimate of this for each spindle, which are probably conservative. However, taking STC's figures for the 19 surviving bearings, the average remaining life was 2.5 years, making a total of 9.5 years. The one failed bearing was an anomaly, as explained in section 6.2.

These results mean that the slower group has an estimated 75% more life time than the faster, and this represents 88% of the total revolutions achieved by the faster. It must also be remembered that when these bearings fail it is by lubrication failure, not fatigue.

Fig.7, as already described in section 6.2, shows the calculated elasto hydrodynamic oil film thickness for the inner race, the inner being the relevant one because it has a thinner film and always shows the most damage. As described in section 6.2, there will be some ball to race contact and also particles in the oil will cause contact. Therefore, to a large extent, the bearings have a component of wear which is proportional to the number of revolutions, since, when there is metal to metal contact, wear particle formation is proportional to distance travelled. Wear particles damage the lubricant by making it into an abrasive mush and possibly also by catalytic action. In other words, slower speed bearings operating under these conditions should last longer, as the experiment confirmed, and clearly this effect would be more marked if the slow speed bearings had the same lubricant film thickness as the high speed ones. The number of slow speed revolutions before failure should roughly equal the number in the high speed case, and the bearing life, under these conditions, can reasonably be regarded as being in inverse proportion to speed, until such time as some other limiting factor appears, possibly lubricant evaporation or creep.

6.7 Cages

(a) Sintered nylon

Forty bearings in groups A, B, D and E had sintered nylon cages located on the single inner land, Figs.3 and 4, and, on the whole, these cages performed very well. The use of a X22 binocular microscope revealed no wear in either the ball pockets or on the inner diameter where it rubs on the inner race land. The contact area is shown by a black stain, but the original machining marks are visible beneath it. It is incredible to think that these surfaces had slid over each other for a distance of 180000 miles without wearing off the machining marks.

Ten bearings were reassembled after the STC examination and the free movement of each cage was similar to that of one bearing fitted with a new cage. However, there was one fault with these cages, which was that they were weak at the ball pockets. There were no broken cages in the 6000rev/min groups, but there were 10 in the 12000. Six of these were associated with mechanical failures of the rigs, e.g. the rotor moving on the shaft to contact the bearing shield and fill the bearing with brass debris, or the preload plunger seizing so that the preload was an unknown value. These 6 failures can be discounted, but the other four showed lubrication failure, although the ball tracks were in good condition. One spindle was running well with a broken cage at the finish. Despite these ancilliary effects, which presumably came first, the excellence of the track condition in some of the cases of failure shows that no great forces were involved and indicates that these cages are too weak. All the breaks were in the walls of the ball pockets.

The ball pockets are tapered and the wall thickness is 0.65mm at the outer surface, 0.95mm at the inner and the cage is 2mm thick radially. The calculated tensile force to fracture one side of the ball pocket is 18N, based on RAE tests of this material, and this is very small compared with the corresponding value for the strip steel cage. There is also the question of the quality control of this porous material, which normally is only about 50% of solid nylon. Presumably the reason for this thin section is that this type of cage has been introduced into an extra light series bearing, designed for a strip metal cage, and space is too limited for a plastic cage of reasonable dimensions. The bearing is 5mm wide and the cage 4.85mm. Another 2mm on the bearing width would have enabled the strength of the cage to have been nearly tripled and the bearing would also have been improved by this modification, since it would increase lubricant capacity and improve alignment.

Sintered nylon also has temperature limitations; it softens at 120°C and cases of balls melting their way through the cage have been known. However, as a storage for oil, this material is much superior to porous phenolic and experience shows that it gives longer life in gyro bearings. The oil storage in cages of this material was 25% to 35% by weight, which is three times as much as contained in cages of the maximum porosity phenolic of the same dimensions. However, it is understood that procurement is difficult and it is to be hoped that the new porous polyimide material will take its place.

(b) Phenolic

The phenolic cages in group G bearings were two piece, with aluminium alloy strengthening rings, which also provided a harder base for the rivet heads (Fig.6). This phenolic was not porous. Fifteen were taken apart by STC and one was left undisturbed. They were all subsequently examined at RAE under a microscope which had an eyepiece graticule, with which measurements down to about 10μm could be made. These cages had parallel sided ball pockets and were outer race piloted. No measurable wear could be seen in any pocket, just a polished area where the ball had touched. There were rub marks on the outer diameter, where contact with the lands had occurred, but in the worse case to be found the depth of wear still could not be measured, i.e. it was less than 10μm. This case was from the only spindle which seized up and it had clearly been hot, as it was black in colour, whereas the others were the normal brown.

In no case had the aluminium rings touched on the lands. The condition overall was excellent.

The porous version of this material can therefore be recommended for long life space bearings and is used in some momentum wheels. It is much stronger than sintered nylon, but absorbs much less oil. It is said that cages made of phenolic give a better performance if they are outer race piloted, as were the samples tested.

(c) Stainless steel strip

These cages in groups C and F deep groove bearings were of two piece design, with the parts fastened together with bent over tabs, and were 'ball riding', i.e. located on the balls (Fig.5). Clearance to the outer race lands was small, about 0.125mm, and at the end of the test slight contact had occurred in about one bearing in six.

All ball pockets showed wear marks at the front and back, the front being in the direction of cage motion. One would expect to find wear in these positions, because the pressure times velocity were greatest there. In addition fore and aft wear in ball pockets can be caused by bearing misalignment but this was not present in these bearings. It is worth noting that all the sintered nylon ball pockets had marks showing fore and aft ball contact also, but the phenolic cages with cylindrical pockets showed front contact only. Contact at front and rear appeared to be connected with the conical shape of the steel and nylon cage pockets.

The wear marks on the stainless steel ball pockets consisted of patches of very highly polished surface which obliterated the normal pickled finish of the metal. The wear particles would be black or dark brown iron oxides; which always appears when steel rubs on itself, whether it be stainless or not, and they tend to be forced into the wedge shaped crevice where the two halves of the cage meet, or into the centre of the pocket where there is little sliding. STC reported thick black polymer at these locations and it is probable that the black was iron oxide. The degree of wear was small, less in depth than could be measured with the eyepiece graticule, i.e. less than 10μm.

The cages performed very well, rather better than was anticipated, and all 20 of the 6000rev/min bearings survived. None of the fastening tabs broke off, a fault which was known to happen to this type of cage some 25 years ago.

However, as the author had previously observed with other experiments, the iron oxide powder this type of cage is prone to produce is abrasive. This no doubt contributed to the fact that the ball tracks were very well worn in comparison to the other bearings and clearly they could not run much longer. While the 10 spindles in the 6000rev/min group were 100% reliable, the two running at 12000 were in hopeless condition, although one was still rotating. About 1 ml of dark brown fretting corrosion products mixed with grease was found in each bearing and in the space next to it.

6.8 Observations on cage stability and torque transients

In the above three types of cage, the wear in all places where sliding occurred was very small. As wear is approximately proportional to force times distance of sliding and the sliding was roughly equal to the distance from the earth to the moon, 180000 miles, the force must have been minute. The relevant coefficients of friction are also small and it therefore follows from this that torque transients due to the cage locking or snatching cannot have occurred to any significant extent. Such transients are important in satellites, either in momentum exchange wheels or despun aerials.

6.9 Power consumption

The rigs were inspected frequently to spot failures and for the first few years the speed run down against time was recorded every 3 months. From the initial slope of this curve, the mechanical power consumption was calculated. One useful result of this procedure was that the only bearing in the 6000rev/min groups to fail could be seen to be in trouble from the beginning, because its power consumption increased continuously.

Nothing else of significance was deduced from these records, which, unfortunately, were not taken right up to the end of the test, at about which time several bearing failures occurred. However, at 30000 hours the average mechanical power consumption of the various groups was as follows:-

Group	No. of spindles	Cage material	Preload N	Speed 1000rev/min	Average mechanical power (watts)
A	10	SN	4.5	6	0.33
B	10	SN	4.5	6	0.40
C	10	Stainless steel	None	6	0.16
D	10	SN	4.5	12	1.13
E	10	SN	4.5	12	1.39
F	2	Stainless steel	None	12	0.57
G	8	Phenolic	4.5	12	0.96

It is interesting that group G, lubricated wholly by grease, required less power than E, which used mostly oil.

7 RECOMMENDED MOMENTUM WHEEL BEARING ARRANGEMENTS

Proposed satellite lives of 10 years are now common and 20 years are talked of[6]. Therefore the lubrication problem is of great importance. This determines the choice of bearing cage material and to some extent the design. The cage encloses and rubs against the balls and is an ideal place to contain oil. It should be porous, non metallic and oil impregnated. There are at least four such materials, sintered nylon, sintered nylon with MoS2 fill, which is much stronger but with less porosity, porous polyimide and porous cotton or linen reinforced phenolic resin. Of these the sintered nylon has the greatest porosity, up to about 30% by weight, and wears excellently. It would seem to be a good choice providing the design is strong, i.e. the ball pocket walls are relatively considerably thicker than the ones tested by the author, and that the running temperature is not more than say 65°C. However, this cage material seems never to have been used in bearings bigger than 6mm bore and therefore it would be inadvisable to put it into a satellite momentum wheel without further experiments for the particular application. Further experiments would also be necessary with the MoS2 filled sintered nylon, although this is being used by the MOD in large bearings, lightly loaded and running at 100rev/min in despun aerial arrangements, and it is working well. It is four times as strong as the ordinary sintered nylon, but its porosity is reduced to about 5% by weight. The polyimide cage material is still being developed, so the choice, for any immediate application, appears to come to porous phenolic resin; because this material has two great advantages, firstly it is the strongest of the four and secondly it has already demonstrated running lifetimes of more than seven years in reaction wheels in space service and also in laboratory tests. However, it is not ideal, it is not 'through' porous and the oil absorption is only about one third that of sintered nylon.

There appears to be a great need for a well developed, easily obtainable, porous plastic cage material. In these experiments, the cages were inner race located, outer race located, and ball riding, and there was nothing to choose between any of these location methods. However, the shapes of the ball pockets in cages which are ball riding sometimes lead to cage instability, due to wedging action. To avoid this possibility, parallel sided pockets (cylinders) are desirable and this means that the cage must be land riding. Whether it rides on the inner or the outer race does not appear to matter, judging by the results of these experiments and others. The bearing manufacturer's recommendation, based on experience, is that porous phenolic cages work better when riding on the outer lands and this view must receive consideration, although the cage then occupies a volume where grease would otherwise collect and form a useful oil creep barrier. Rivets can lead to trouble and therefore the cage must be of one piece construction.

The bearings should clearly be angular contact and separable for cleaning and impregnation purposes. They should be cold temperature stabilized as well as high temperature stabilized. The inner race should be the one with one land cut away, cutting away an outer land simply invites the precious lubricant to run out. The contact angle should be about 18° with a small tolerance, as high angles, like 30°, cause excessive ball spin and low angles, like 10°, cause high resultant ball load when subject to axial preload or distortions.

It is normal for space purposes to use the 'extra light' series of bearings, presumably because they are very light in weight and save housing weight. However, they are also light in construction and in small sizes have insufficient space for a good cage or store of grease. The alignment or seating of the narrow, sliding, outer race is not good and thicker rings would be more dimensionally stable. Using double width bearings, as described in this Report, is one solution to this problem, but they are not available in a wide range of sizes. It is therefore suggested that the more robust 'light' series should be considered, which, incidentally, are so much greater in load capacity that a smaller diameter may be suitable. For example, a 15mm bore 'extra light' 12° contact angle bearing has 280kgf static load capacity, whereas the corresponding 'light' bearing of only 10mm bore has a static load capacity of 302kgf.

The bearing housing arrangements should be designed in accordance with the principles outlined in sections 2.7, 2.8 and 2.9. If it is possible to use a diaphragm spring for axial preload, so as to avoid sliding parts, this should be done; but even if it is not, it should be possible to avoid fretting, since most of the test assemblies in the experiments reported here did not fret and the preload system which was used is a common commercial method. It is not necessary to prevent free movement of the bearings or other parts during vibration. The author has experience of several hundred bearing spindles which were allowed to rattle during vibration testing and no damage could be discovered, either by 'smoothrator' testing of the bearings or by subsequent performance. The vibration force used in these other tests was up to one third of the static load capacity of the bearings. Thus many bearing mechanisms have been successfully laboratory tested and others, also using compliant preload systems, have given excellent performances in space. During launch vibration, the parts are allowed to rattle against fixed stops and particle formation has never been a problem. The compliant preload method makes assembly easier and therefore more reliable. The low preload recommended for space use should be just sufficient to cause the balls to rotate without skidding and is best found experimentally.

Some momentum wheels now being made use solid preloading. Manufacturers claim that dimensional accuracy is now such that preload variations due to errors are adequately small, whereas this was not so when these ball bearing life tests began. Such a design may, therefore, be viable, but it will require the most careful consideration of temperature gradients as well as of bearing accuracy. The common argument for this arrangement is that it avoids resonances at frequencies which could be in awkward bands of the vibration spectrum of the satellite during launch.

From the author's experience of bearing steels stainless should be used if it is available, but if not, there is no need for concern. Obviously the best grade of steel available and high accuracy bearings, such as ABEC7 grade, should be chosen.

The oil viscosity should be such as to separate the balls from the races by a reasonable elasto hydrodynamic film thickness, say 0.25μm. The oil temperature should be, ideally about laboratory temperature, or less; the bottom limit is the pour point.

8 LIFE PREDICTION

In this tentative life prediction for momentum wheel bearings it is assumed that fatigue, lubricant evaporation, creep, and chemical change have been arranged so as not to be limiting factors. Fatigue can be eliminated by the adequate sizing of bearings, a low vapour pressure oil will cope with evaporation dangers, creep is no problem, if anticreep barriers or some grease are used, and mineral oil is everlasting in an inert environment. Chemical degradation of mineral oil can be made negligible by avoiding high temperatures and using an inert gas fill. The bearing will then fail when the lubricant fails and it is postulated that this occurs at a given level of particule contamination. Wear particles build up an abrasive mush and can also interact chemically with the lubricant. In the following arguments rough approximations only are intended.

There is no pure rolling, especially in an angular contact bearing, so wear particles are produced in a ball race by sliding between the balls and raceways, and between the balls and the cage, even though there is a separating film of oil. A possible explanation is that contaminant particles which were there originally bridge the gap provided by the oil film. The rate of particle formation is a product of the load and the sliding distance, and it is independent of the apparent contact area. In support of this statement, section 6.6 shows that bearings of the same type and with the same load had a life which was inversely proportional to their speeds, provided that the oil film thickness conditions were the same at the different speeds.

If the effect of a change of load, which is equivalent to change of diameter, could also be assessed, a more universal formula for estimating the life of bearings in momentum wheels could be postulated.

All components for satellites have to be tested on earth and the bearings for momentum wheels will need to have a greater load capacity than the ones with which this Report deals. This could be achieved by increasing the stress in the bearings, or by increasing their size; that is by increasing the number and diameter of the balls without changing the stress. In fact, it is not possible to change the stress much, as the following argument shows. The relevant fatigue life is the L1 or 0.99 reliability case, which, when applied to groups D and E, yields a figure of 70 years. Fatigue life varies very sharply with stress, in fact inversely with the ninth power, for a given bearing, and it is reasonable to expect similar stress levels to apply to other size bearings for a given fatigue life. So clearly the value cannot change much from the $620 MN/m^2$ (90000psi) which occurred in groups A, B, D, E. It therefore follows that increased load must be catered for by bearings of larger diameter and width.

The dynamic load capacity of bearings using given sized balls varies with diameter, because the number of balls also depends linearly on diameter. However, when the ball size is changed the slope of the graph of load against diameter changes for a further range of bearing sizes. One bearing catalogue showed that the load capacity of the range in question was linear from 10 to 40mm bore. Dealing first with this situation:-

Particle formation $\propto$ LS — where L = load, S = sliding distance, D = bearing diameter

but $L \propto D$

and $S \propto D$ for a given rotation speed

therefore particle formation $\propto D^2$.

But the storage of lubricant in a bearing increases also at least with D^2, as the cage is longer and it and the lands are usually made wider as diameter increases. Therefore the ratio of particles to lubricant is no worse in a large bearing than in a small one.

A similar result is obtained if load is increased by increasing ball size only, as load per ball varies with the square of ball diameter. For constant stress and fatigue life the bearing diameter does not increase proportionally to load increase, so sliding distance for a given load is reduced, but cage width and radial thickness are both increased and so lubricant storage is increased even more favourably.

On this basis, variation in load which is accommodated by variation in bearing size does not change life significnatly. Therefore, if a momentum or reaction wheel is run at 2000rev/min with bearings which are made to the foregoing principles, its life would be three times that of the 6000rev/min groups, i.e. 27.5 years. In this it is assumed that equivalent bearings and lubricant are used, whereas better versions of both are now available. Also, applying current knowledge of elasto-hydrodynamic lubrication would provide a further bonus. Support for this theory is derived from a survey of the practical experience given in unpublished RAE work.

If this theory is applied to 100rev/min despun aerial bearings, the predicted life is so long that one hesitates to print it. However, such bearings are exposed to the vacuum of space, so the lubricant reservoirs for vapour lubrication would run out after about 50 years.

9 EXAMPLES OF LONG LIFE IN BALL BEARINGS IN COMMERCIAL EQUIPMENT

A computer memory disc, in Space Department RAE, provides some supporting data on the life capabilities of larger bearings. Since stopping and starting is the possible unreliabile operating mode for the flying 'read write' heads, this disc is kept running continuously. It has now done 7.3 years at 1745rev/min with no bearing troubles whatever and no relubrication. The disc is 0.67m diameter, weighs about 12kg, and is supported on an overhung shaft with a horizontal axis. Its momentum of about 120N m s is considerably larger than is usually used in satellites and it does not have the advantage of a weightless environment. The bearings are ball races, inner races rotating, grease lubricated and compliantly preloaded, but the lubricant is unknown. The size, taken from the inner diameter, is about ten times as big as the small bearings of this Report. These bearings - in standard commercial equipment, demonstrate what can be done in the larger sizes, in spite of the fact that some primary long life factors are missing, such as an inert gas atmosphere and porous cages. A second version of this mechanism has currently completed three years satisfactorily.

10 CONCLUSION

Ample evidence has been obtained that a reliable life of 10 years for a ball race type of bearing for a spacecraft momentum or reaction wheel is quite feasible. Twenty years is probable.

To achieve this performance the main requirements are:-

Low operating and storage temperature.

Low preload, preferably applied by a compliant diaphragm type spring.

Low speed.

Inner race rotation.

A porous cage, soaked with a low vapour pressure oil.

A small amount of grease.

The lubrication arrangements to be such as to provide an elasto-hydrodynamic film of about 0.25μm thickness.

An inert gas atmosphere, with the bearing sealed against the passage of gas.

Angular contact precision bearings, including a low tolerance on the contact angle.

Great care in the design to ensure very good mechanical alignments.

A high standard of cleanliness.

Table 1

GENERAL DETAILS OF TEST BEARINGS AND THEIR OPERATION

Bearing Group	A	B	C	D	E	F	G
Rig Nos.	1–10	11–20	21–30	31–40	41–50	51 and 52	53–60
Speed (rev/min)	6000	6000	6000	12000	12000	12000	12000
Type No.	1	2	3	1	2	4	5
Size	5 × 16 × 5mm	5 × 16 × 5mm	5 × 16 × 5mm	5 × 16 × 5mm	5 × 16 × 5mm	5 × 16 × 6mm	5 × 16 × 10mm
Description	AC	AC	DG	AC	AC	DG	DG
Cage	SN	SN	SS	SN	SN	SS	P
Cage location	inner race	inner race	balls	inner race	inner race	balls	outer race
Contact angle	$17–27^{o}$	$17–27^{o}$	-	$17–27^{o}$	$17–27^{o}$	-	6^{o}
Diametral clearance	-	-	8–13μm			6–10μm	
Preload	4.5N	4.5N	-	4.5N	4.5N	-	4.5N
Average amount of oil in cage (mg)	44	43	-	55	14	-	-
Average amount of grease in bearing (mg)	10	10	113	10	10	121	210
Average mech power (watts)	0.3	0.3	1.1	1.1	1.1	0.6	0.8

NOTE: Type 1 bearings have inner races integral with the shaft, all others are normal, Figs.4, 5, 6 and 7. All bearings have 3.1mm (0.125in) balls and are made of EN 31 type carbon chrome steel.

AC = angular contact
DG = deep groove
SN = sintered nylon
SS = stainless steel
P = phenolic

Table 2

SUMMARY OF BEARING TEST RESULTS

Group	Brief data	Overall conclusion
A	9 spindles ran 61180h test. One failed at 36600h. This is thought to be due to human error. Inner track condition:- 3 VG, 5 G, 7 F, 3 P. Plenty of oil and grease left on bearings.	Good
B	All 10 spindles ran 61180h test. Inner track condition: 2 VG, 16 G, 2 F. Plenty of oil and grease left on bearings.	Very good. The best performer
C	All 10 spindles ran 61180h test. All bearings are heavily worn by visual observation and Talysurf. Tracks very wide. Grease generally black with iron oxide.	Despite the wear, 100% reliable
D	Test 54580h. All spindles exceeded 50000h. 4 rig failures due to contact between rotor and bearing; these yielded no information. 2 bearing failures at about 53000h, apparently due to lubrication failure followed by cage fracture. Four surviving spindles, bearings 5 G, 2 F, 1 P. Surviving bearings dry and apparently near end of life.	Valuable information from survivors, see text
E	Test 54580h. 4 rigs taken off test at 31000h because preload plungers had seized. Two bearing failures, one at 33000h which may have been due to excessive load caused by sliding bearing being very stiff in housing, and one at 53900h - apparently lubricant failure followed by fracture of both cages. 4 surviving spindles, bearings 5 G, 3 P. Surviving bearings dry and near end of life.	Comment as D
F	These bearings were in hopeless condition, full of fretting corrosion powder and about 1 ml of this was in the space beside the bearing. One spindle still running at end of 54580h test.	Past the limit for this type
G	7 out of 8 spindles completed 54580h test. One failed at 51000h. Apparently lubricant failure. All tracks showed some pitting and one bearing had severe fatigue spall on one inner race and one ball. This fatigue damage caused by high resultant load due to very low contact angle. If this had been normal, the performance would have been very good. Ample grease in good condition in the survivors.	Could have been the best overall performance

VG and G = very good and good. With X22 binocular microscope it can be seen that the track is smoother than the original finish but no texture is visible.

F and P = fair and poor. Some etched surface can just be seen with X22 microscope.

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Reports quoted above are not necessarily available to members of the public or commercial organisations.

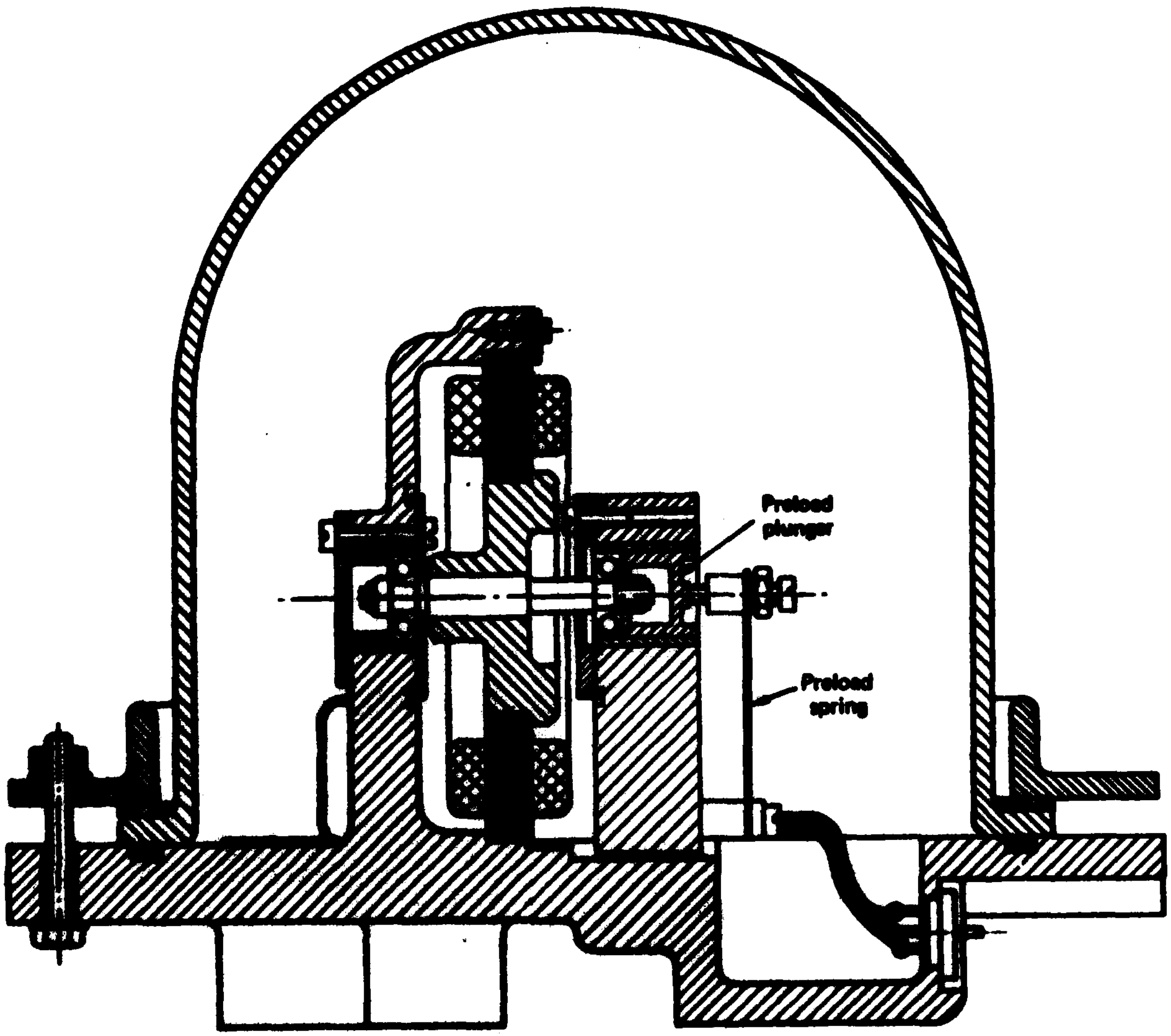

Fig.1 Bearing test rig

Fig.2

Fig.2 Bearing test rig

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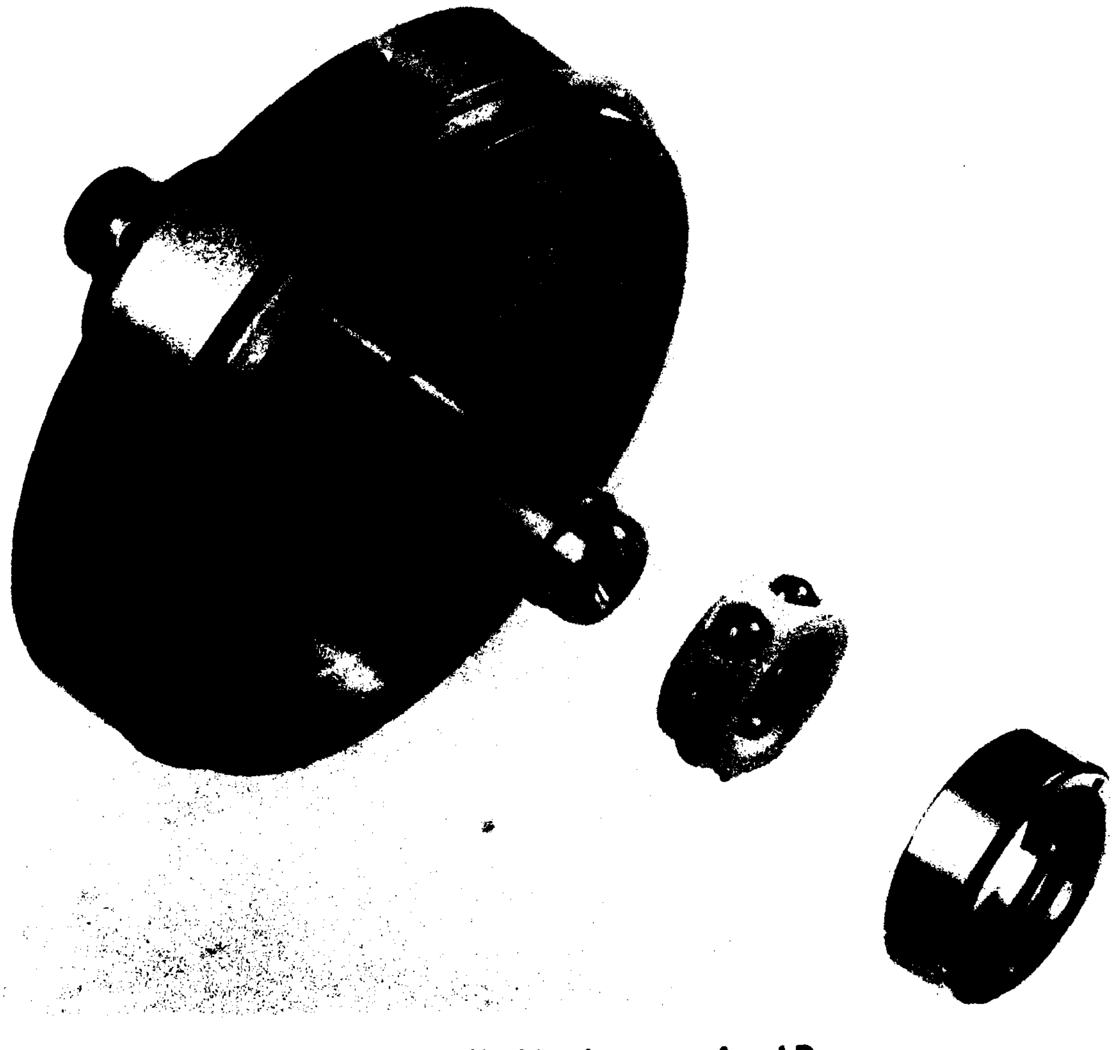

Fig.3 Used bearing groups A and D

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Fig.4

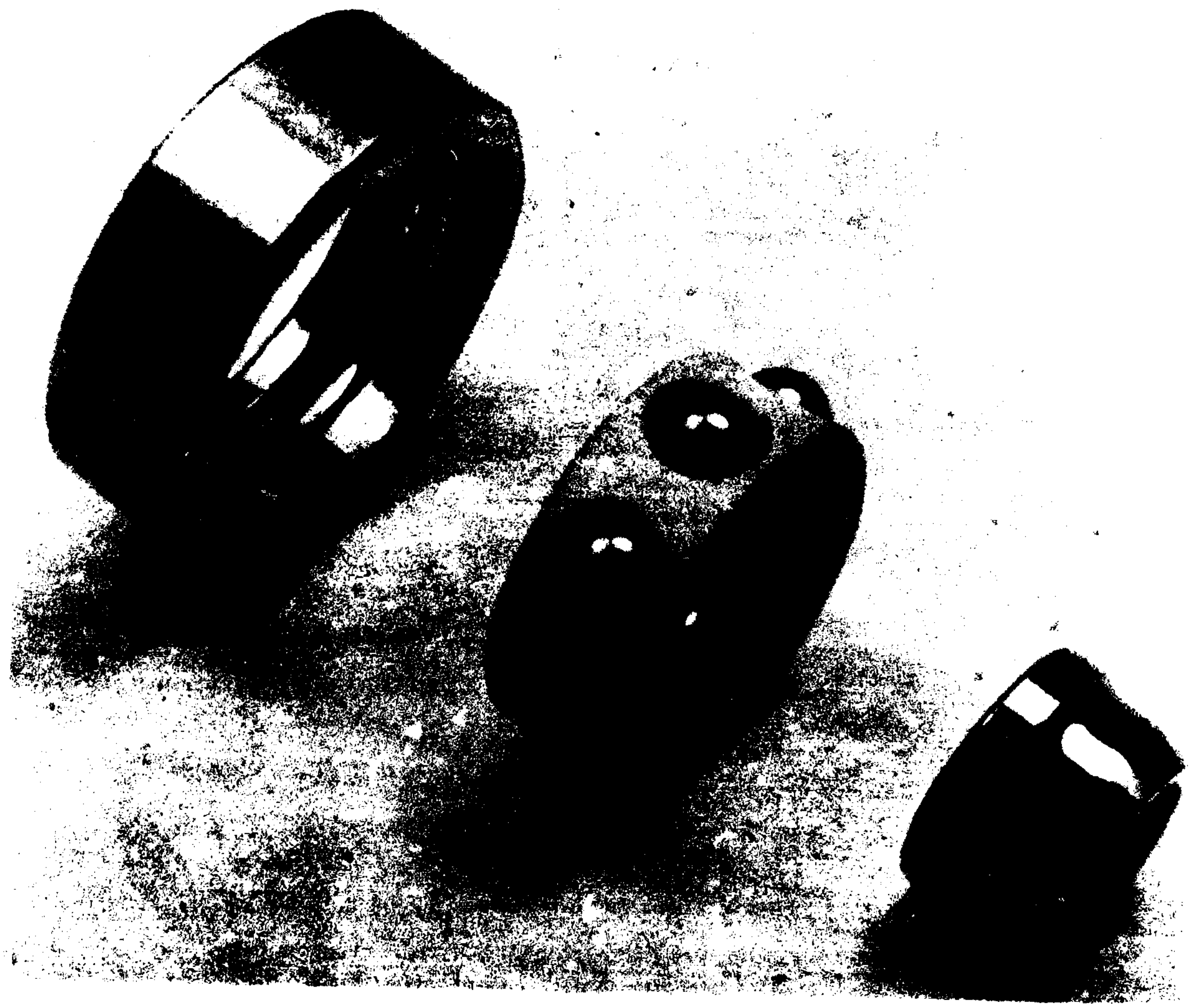

Fig.4 Used bearing groups B and E

Fig.5 Used bearing group C

Fig.6

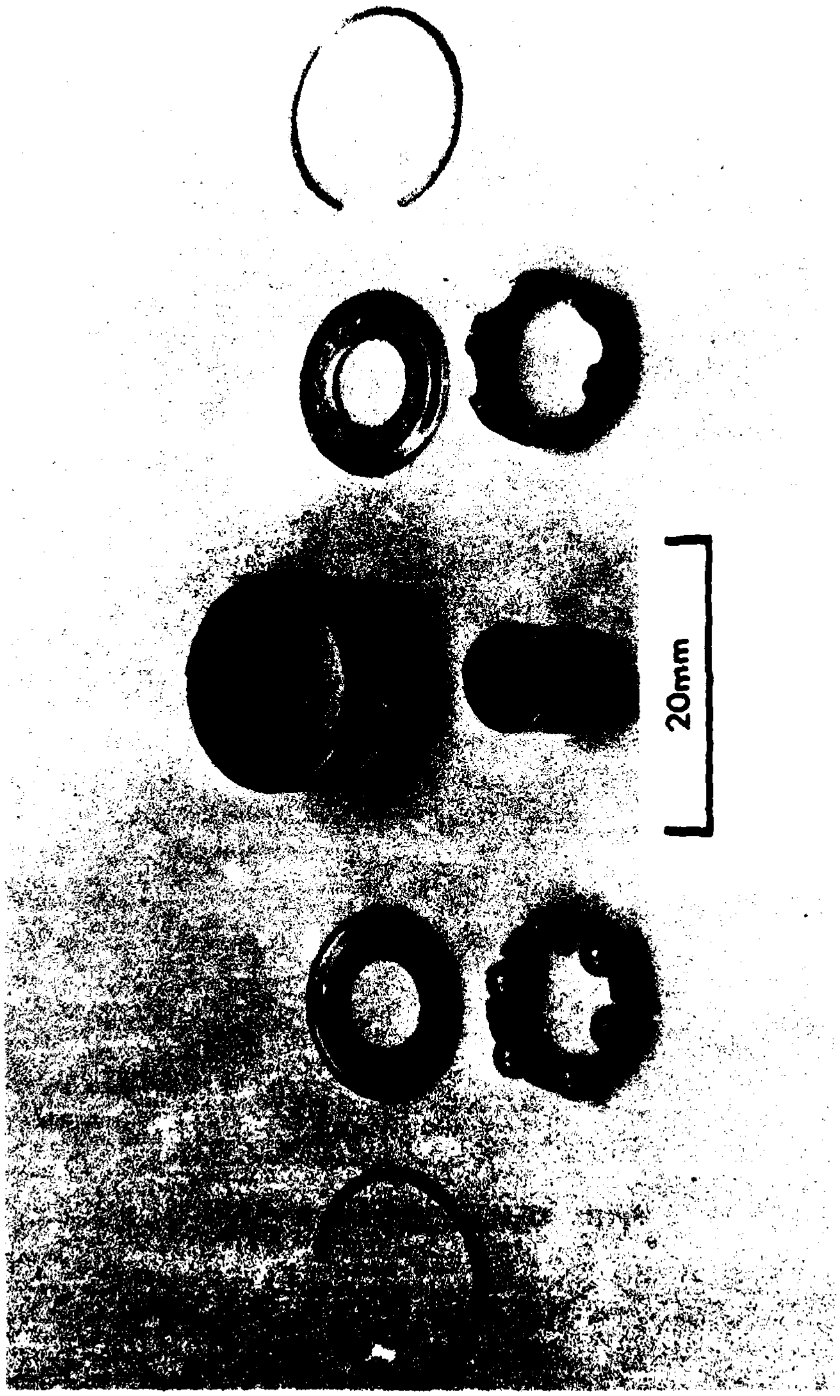

Fig.6 Used group G bearing

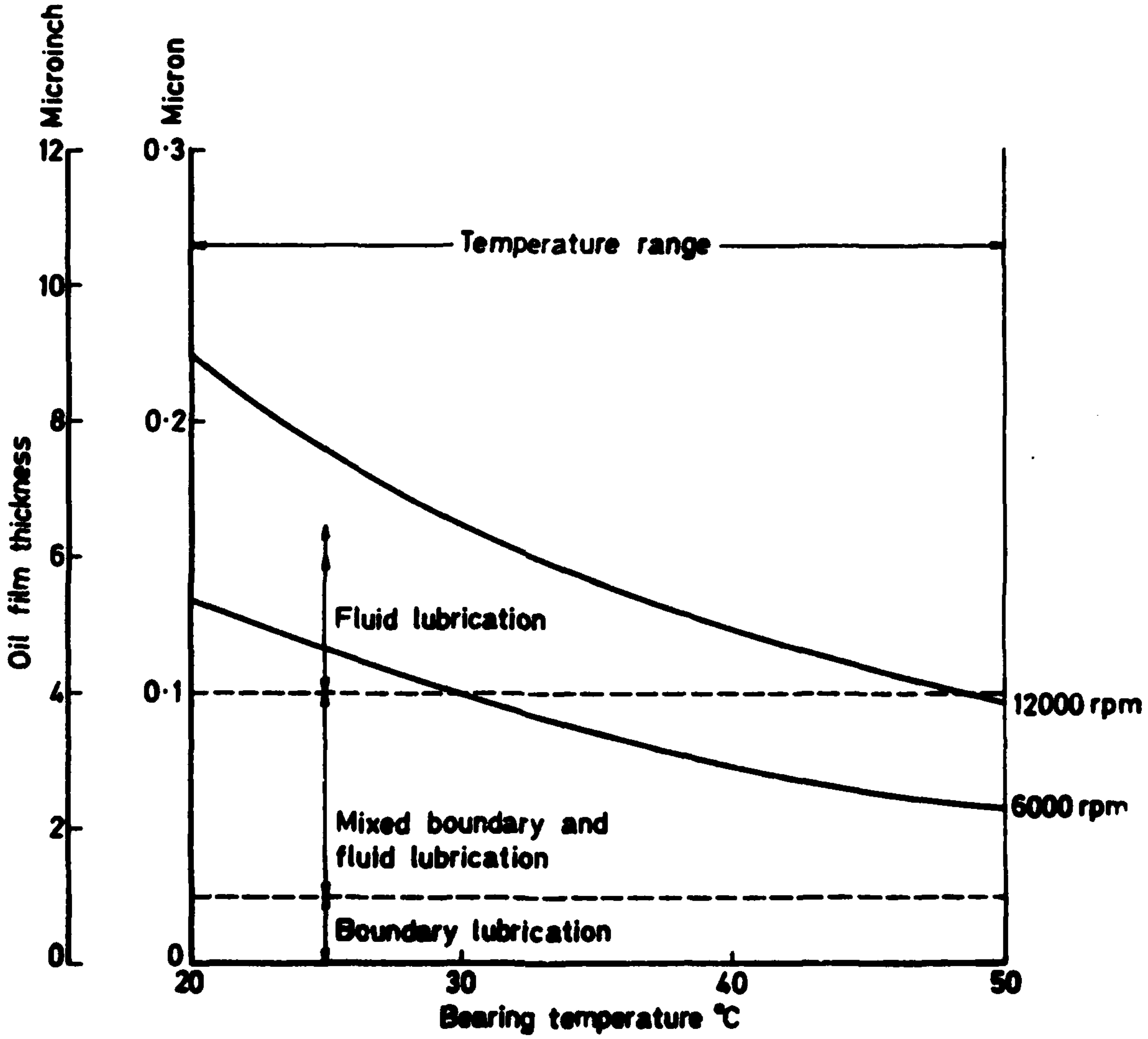

Fig.7 Lubrication regimes, inner races

REPORT DOCUMENTATION PAGE

Overall security classification of this page

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17. Abstract

The Report concerns the development of long life ball bearings with one initial lubrication. The knowledge of many experts was pooled and an experimental batch of 60 spindles fitted with several types of ball races was run for seven years. The conclusion is that a 10 year life should be reliably obtained for such devices as momentum or reaction wheels for satellite attitude control. Significantly longer reliable lines are probably possible.